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# EFFECTS OF ROUGHNESS ON HEAT TRANSFER IN CONICAL NOZZLES

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# EFFECTS OF ROUGHNESS ON HEAT TRANSFER IN CONICAL NOZZLES

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ABSTRACT

The changes in the nozzle wall surface finish due to roughness may affect the gas to wall heat transfer especially if the roughness peaks protrude through the sublayer of the boundary layer. This paper represents a study on the effects of various degrees of surface roughness on heat transfer in two conical converging diverging nozzles with different convergence angles using heated air. Roughness levels up to 325 rms were considered. Roughness causes detransition from turbulent flow to take place at a lower Reynolds number than that for a smooth wall. In the turbulent regime the heat transfer is not noticeably affected until the roughness height is greater than a given height which seems to correspond to the approximated sublayer height.

NOMENCLATURE

| | |
|---|---|
| A | nozzle cross-sectional area |
| $C_f$ | local skin friction coefficient |
| $C_1, C_2, C_3$ | constants |
| D | local nozzle diameter |
| h | local heat transfer coefficient |
| i | enthalpy |
| Pr | Prandtl number |
| p | pressure |
| q | heat flux |
| Re | Reynolds number |
| rms | root mean square in micro-inches |
| T | total temperature |
| t | static temperature |
| u | velocity |
| $u^+$ | non-dimensional velocity |
| x | axial coordinate measured from nozzle throat |
| Y | distance along heat-flux meter measured from gas-side wall |
| y | distance normal to the wall |
| $y^+$ | non-dimensional distance normal to the wall |
| $\beta$ | angular position of nozzle instrumentation |
| $\mu$ | dynamic viscosity |
| $\rho$ | gas density |

Subscripts:

| | |
|---|---|
| ad | adiabatic wall condition |
| D | based on diameter |
| i | based on enthalpy |
| m | on heat flux meter |
| o | stagnation condition |
| ref | reference condition |
| s | static condition |
| t | throat |
| w | wall condition |
| $\infty$ | free stream condition |

INTRODUCTION

Recent studies have been conducted on the gas-to-wall heat transfer of accelerated flows in smooth nozzles with different convergence angles. These studies have shown that the heat transfer in accelerated flow differs considerably from that predicted by unaccelerated flow theory and experiment.[1-4] However little consideration was given to the effects of the wall surface finish on the nozzle heat transfer. It is therefore necessary to assess the effect of surface roughness on nozzle heat transfer.



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Acceleration causes a thinning of the boundary layer whereas roughness causes an increase in the surface height. Therefore the changes in the wall surface finish due to roughness may affect the gas-to-wall heat transfer especially if the roughness peaks protrude through the sublayer of the boundary layer. Applying this to chemical rockets, it is found that after long periods of operation at high temperatures the walls of most nozzles become rougher due to erosion and oxidation. These differences in surface texture introduce questions concerning the applicability of the smooth wall heat transfer results to a rocket nozzle. Another source of roughness in certain rocket nozzles is a ceramic coating which provides insulation between the hot propellant-gas and the cooled wall. A knowledge of the effects of roughness on the heat transfer is desirable in order to determine whether the reduction due to the insulation is offset by an increase in heat transfer resulting from the rougher surface.

In order to obtain an understanding of the heat transfer effects of roughness in accelerated flow one must first consider independently the behavior of heat transfer in nozzles (accelerated flow) with smooth walls. The heat transfer studies of Refs. (1) to (4) have shown that the heat transfer in accelerated flow differs considerably from that predicted by unaccelerated flow theory and experiment. In accelerated flow there are two distinct regimes of heat transfer rate and for both the heat transfer is less than that of conventional turbulent pipe flow. The high Reynolds number regime represents turbulent heat transfer for accelerated flows and the low Reynolds number regime suggests a laminarization phenomenon (sometimes called "detransition"). The laminarization depends on the flow acceleration and is further discussed in Ref. (5).

This paper represents an experimental study on the effects of various degrees of surface roughness on heat transfer in two converging-diverging nozzles. A $60^{o}$ half-angle of convergence nozzle is compared to a $30^{o}$ half-angle of convergence nozzle for acceleration and roughness effects on the gas to wall heat transfer characteristics. Both nozzles have a divergence half-angle of $15^{o}$. The nozzles were roughened by a sandblasting technique to roughness levels up to 325 rms. Air at a stagnation temperature of $970^{o}$ R was used as the working fluid. The pressure was varied from 30 to 300 psia yielding almost an order of magnitude range in Reynolds number at each station. The Reynolds number range at the throat station was $6\times10^{5}$ to $5\times10^{6}$. These operating conditions made it possible to obtain heat transfer coefficients and parameters in the turbulent, transition and laminarization regimes.

## APPARATUS

The experimental apparatus, shown in Fig. 1, comprised a heat exchanger, diffuser, plenum, pipe inlet, nozzle, and exhaust system. Uncooled- and cooled-wall pipe inlets having inside diameters of 6.5 in. were coupled to $30^{o}$ and $60^{o}$ half-angle of convergence by $15^{o}$ half-angle of divergence water cooled nozzles. These nozzles will hereinafter be referred to as simply the $30^{o}$ and $60^{o}$ nozzles.

The adiabatic and cooled inlets have total lengths of 17.0 and 37.6 in., respectively. A boundary layer was initiated at the leading edge of the inlets by means of a bleed flow arrangement. Based on the Reynolds number level and surveys in the inlets, the velocity boundary layer appeared to become turbulent very close to the leading edge. In the cooled inlet, the thermal boundary layer developed over a length of 24.2 in. In tests with the adiabatic inlet, the thermal boundary layer started to develop at the nozzle entrance.

The water-cooled $30^{o}$ and $60^{o}$ nozzle configurations along with tables of the instrumentation sites and coordinates are presented in Figs. 2 and 3, respectively. Each nozzle had a nominal radius of curvature and throat diameter of 1.5 in. thus providing a contraction area ratio of about 18.8. The expansion ratios for the $30^{o}$ and $60^{o}$ nozzles were 25.3 (Mach 5.0) and 3.32 (Mach 2.7), respectively.

## MEASUREMENTS

Local heat transfer rates and wall static pressures in the nozzles were measured at the stations listed in Figs. 2 and 3. The heat transfer rates and wall temperatures were determined from temperature measurements on Inconel plug-type wall heat flux meters which have been described in Refs. (2) and (5). Ratios of static-to-total pressure at each station have been presented for smooth wall tests in Ref. (2). In this study the effect of roughness on the measured static pressures was determined to be negligible; therefore the pressure ratios presented in Ref. (2) are applicable to the present results.

## METHOD OF ROUGHENING

In this study we are interested in a "natural" or "uniform" type of roughness where the peaks and valleys are connected by a gradual slope rather than a vertical face. Nozzle wall roughness values up to 325 rms were obtained by a sand or grit blasting technique. The range of roughness levels was obtained by varying the size and material of the abrasive, and the blasting pressure. This method left a naturally rough surface with minimal damage to the instrumentation. The surface was measured with a roughness meter giving a root-mean-square value of surface height expressed in micro-inches.

## DATA REDUCTION

The local heat transfer rate $q$ was computed from the observed temperature gradient in the heat-flux meters. This temperature gradient is described by the Fourier conduction equation which can be integrated over the length of the heat meters to give

$$-qY = C_1 t_m^2 + C_2 t_m + C_3 \tag{1}$$

The constants $C_1$ and $C_2$ were determined from a thermal-conductivity calibration of the heat meter material. The values of $q$ and $C_3$ were obtained from the measured temperature on the heat meter, $t_m$, and the corresponding thermocouple location $Y$ at two positions on the meter. The wall temperature was calculated by setting $Y = 0$ which corresponds to the wall position.

The errors associated with the heat flux measurements for smooth walls have been discussed in Ref. (5). These errors were expected to be within ±10 percent of the measured heat transfer rates. This error is assumed to apply to the present results for rough walls.

The heat transfer results will be presented in terms of a heat transfer coefficient $h_i$ and also as a non-dimensional grouping $St_{ref}Pr^{0.7}$. The heat transfer coefficient is given by

$$h_i = \frac{q}{i_{ad} - i_w} \tag{2}$$

where the adiabatic enthalpy was calculated from

$$i_{ad} = i_s + Pr^{1/3}(i_o - i_s) \tag{3}$$

The non-dimensional heat transfer grouping $St_{ref}Pr^{0.7}$ is given by

$$St_{ref}Pr^{0.7} = \frac{h_i}{\rho_{ref}u_\infty} Pr^{0.7} \tag{4}$$

The $St_{ref}Pr^{0.7}$ grouping will be presented as a function of the Reynolds number given by

$$Re_{D,ref} = \frac{\rho_{ref}u_\infty D}{\mu_{ref}} \tag{5}$$

A Prandtl number of 0.71 was assumed in the above equations. The subscript "ref" denotes that properties were evaluated at a reference enthalpy condition given by

$$i_{ref} = i_s + 0.5(i_w - i_s) + 0.22\ Pr^{1/3}(i_o - i_s) \tag{6}$$

## RESULTS AND DISCUSSION

In studying the effects of roughness on heat transfer in conical nozzles undergoing different rates of acceleration both a 60° and 30° nozzle were used. The results for the 60° nozzle will be presented in a brief form since they were reported fully in Ref. (6) whereas the results for the 30° nozzle will be treated more thoroughly.

### Heat Transfer in the 60° Nozzle

A summary of the heat transfer results for the 60° nozzle is found in Fig. 4. Heat transfer in the form of a Stanton Prandtl number grouping $St_{ref}Pr^{0.7}$ is plotted with respect to Reynolds number at the throat station for both a cooled and an adiabatic inlet. The advantage of this particular non-dimensional presentation is that it shows the laminarized and turbulent modes of heat transfer as distinct regions which can be identified by a Reynolds number in which the reference dimension is the local diameter. The diameter is chosen as a convenient dimension since it is approximately proportional to the boundary layer thickness which, of course, would be a more appropriate dimension if it were known (e.g., ref. (3)). Therefore it must be remembered that two points at the same Reynolds number based on diameter, but at different stations, will not necessarily have the same heat transfer. The throat station is presented because it is the region one is most concerned with in a nozzle and the heat transfer is typical of accelerated flow. Both a cooled and an adiabatic inlet each having smooth walls, were used in order to determine the effects of the thermal history of the fluid on the nozzle heat transfer.

Besides the experimental heat transfer data, Fig. 4 also shows the standard pipe-flow type of nozzle correlations. The turbulent correlation

$$St_{ref}Pr^{0.7} = 0.026\ Re_{D,ref}^{-0.2} \tag{7}$$

and the laminar correlation

$$St_{ref}Pr^{0.7} = 0.29\ Re_{D,ref}^{-0.5} \tag{8}$$

frame the results very well. Although there now are much better methods of predicting nozzle heat transfer these correlations represent the upper and lower limits of the experimental results and will serve as reference levels. However, they show no comprehension of acceleration, fluid thermal history or surface roughness.

The smooth nozzle was successively roughened three times and the resulting heat transfer parameters using a cooled inlet are shown in Fig. 4(a). Two separate regimes of heat transfer emerge with a third regime connecting the two, as was also observed by the authors of Refs. (3) to (5). The mode of heat transfer in the upper regime is turbulent. In the lower regime the heat transfer reflects a reduction in turbulence commonly called laminarization. Although the heat transfer may be close to the values based on the correlation for a laminar boundary layer there is reason to believe the structure of the thermal boundary layer remains turbulent for the total plenum pressures used in this report.[5] The reason being that the temperature profiles from Ref. (5) were more turbulent-like than laminar. The heat transfer regime connecting the upper and the lower corresponds to a transition phenomenon.

In the turbulence and transition regimes an increase in roughness height causes an increase in heat transfer rate. At the highest Reynolds number, $Re_{D,ref}$, all the experimental values of heat transfer parameter are in the turbulent regime. At the first level of roughness the increase in heat transfer is negligible (less than 10 percent), at the second level it increases by 17 percent and at the third level the increase is 30 percent. As the Reynolds number is decreased turbulent flow undergoes detransition. Roughness prolongs the regime of turbulent heat transfer rate and causes detransition to take place at a lower Reynolds number than it normally would for a smooth wall. At the low Reynolds numbers the smooth and the 120 rms wall are in the laminarization regime where the effect of roughness on heat transfer is negligible.

In Fig. 4(b) the heat transfer results for the adiabatic inlet are presented. They are similar to those of the cooled inlet except that for a given Reynolds number the corresponding value of $St_{ref}Pr^{0.7}$ is higher. The increase in $St_{ref}Pr^{0.7}$ is 50 and 25 percent for the turbulent and laminarized regimes respectively. With the cooled inlet the thermal boundary layer begins to grow in the inlet and continues growing in the nozzle. With the adiabatic inlet the thermal boundary layer does not exist in the inlet and starts growing at the entrance of the nozzle. This means that at all locations in the nozzle the thermal boundary layer produced using the cooled inlet is thicker than that using the adiabatic inlet.[3] The turbulent boundary layer can be thought of as a conduction layer which transmits or conducts heat through it. As a conduction layer, its transmissivity decreases with increasing thickness of the layer. Therefore the nozzle heat transfer for tests with the cooled inlet is less than the values corresponding to tests with the adiabatic inlet.

In working with pipe flow Nikuradse[7] found that in the turbulent regime the effects of roughness become significant when the roughness height is greater than what is often referred to as the sublayer height. In Ref. (6), a $u^+$, $y^+$ model of the boundary layer profile was used to estimate the sublayer height for the $60^o$ nozzle. The sublayer height, y, was obtained from the following relation:

$$y = \frac{y^+ \, D \, \frac{\mu_w}{\mu_\infty}}{\sqrt{\left(\frac{C_f}{2}\right)_{ROUGH} \frac{t_\infty}{t_w} Re_D}} \qquad (9)$$

It is recognized that the sublayer height is a quantity which is not well defined; however the sublayer is usually considered to be the region in which the average viscous stress is nearly constant. In Eq. (9) this region was assumed to be given by $y^+ < 20$.

In the turbulent regime the heat transfer was not noticeably affected until the roughness height was in the region of or greater than the approximated sublayer height. Although good agreement was obtained with the Nikuradse hypothesis this conclusion should not be emphasized until either actual measurements or very accurate calculations are made to find the sublayer height.

## Heat Transfer in the $30^o$ Nozzle

In Fig. 5, the heat transfer coefficient, $h_i$, is presented as a function of axial distance from the nozzle throat for a given plenum pressure with the nozzle roughness as a parameter. This form of presentation shows the direct magnitude change in heat transfer coefficient and its axial distribution, which is not clear from the previous figures of $St_{ref}Pr^{0.7}$ vs. $Re_{D,\,ref}$. The plenum pressures that are used are the same as for the $60^o$ nozzle reported in Ref. (6). For the $30^o$ nozzle the range of roughness levels was extended to include a value of 55 rms.

In Fig. 5(a), at a stagnation pressure, $p_o$, of 300 psia, the heat transfer coefficient reaches a maximum in the vicinity of the throat and then decreases as the flow goes supersonic. For a stagnation pressure, $p_o$, of 75 psia, Fig. 5(b) shows the same general axial heat transfer distribution except that at the stations just upstream of the throat there are large differences in the heat transfer coefficient between the smooth wall and the 180 and 325 rms walls. At station 9 the heat transfer of the smooth and 55 rms walls are in the lower portion of the transition regime while the heat transfer for the two rough walls are turbulent. In Figs. 5(b) and (c) the heat transfer reaches a local minimum at station 14 then reverses itself and reaches a local maximum at station 15. This same phenomenon was observed for the $60^o$ nozzle[6] and also in Ref. (4) where the investigator used a smooth nozzle wall and varied the stagnation pressure. In all these cases the increase in heat transfer occurred where the nozzle exit cone meets the throat radius of curvature and this change in geometry causes a slight adverse pressure gradient.

In pipe flow there are cases where the surface may seem rough to the senses of sight and feel but behaves as a smooth surface.[8] The rough pipe is then said to be hydraulically smooth. Similarly, we find in Fig. 5 that at almost all the nozzle stations the heat transfer coefficient for the 55 rms wall is the same as that for a smooth wall. Therefore, one can assume that there is a specific roughness value below which a nozzle can be considered hydraulically smooth.

Experimental heat transfer parameters at various stations in the $30^o$ and $60^o$ nozzles are presented in Fig. 6. The plenum pressure is varied in order to get different flow rates which in turn give a range of Reynolds numbers at each station. The heat transfer parameter, $St_{ref}Pr^{0.7}$, at each station is presented as a function of Reynolds number with the nozzle roughness acting as a parameter. The variation of $St_{ref}Pr^{0.7}$ at stations 9, 11, and 14 for the $30^o$ nozzle are shown since they represent significant locations in the subsonic, sonic and supersonic regimes of flow. As expected the heat transfer results fall into regimes of turbulence, transition and laminarization. The heat transfer distributions are very similar to those of the $60^o$ nozzle of Ref. (6) and will be compared in the following section.

## Comparison of the Heat Transfer in the $30^o$ and $60^o$ Nozzles

In Fig. 6 the heat transfer in the two nozzles are compared at stations which have nearly the same local nozzle diameter. As mentioned in the beginning of this section on results and discussion, the local nozzle diameter is chosen as the characteristic dimension in the Reynolds number rather than the more appropriate dimension of boundary layer thickness. Therefore, it must be remembered when comparing the results for the $30^o$ and $60^o$ nozzles at a given station, that two points at the same Reynolds number based on diameter do not in theory have the same Reynolds number when boundary layer thickness is used as a characteristic dimension. However, since the appropriate boundary layer thickness Reynolds number is an unknown quantity the results are presented in terms of a measurable quantity; namely, the Reynolds number based on diameter.

The increases in the turbulent regime heat transfer parameter with respect to the smooth wall for the $60^o$ nozzle are 10, 17, and 30 percent for the respective roughness values of 120, 180, and 325 rms. Due to the poor condition of some of the inaccessible thermocouples in the $30^o$ nozzle, the increases in heat transfer due to roughness are not as consistent as those from the $60^o$ nozzle nor are they consistent with each other from station to station. For both nozzles, roughness causes transition from laminarized flow to take place at a lower Reynolds number than that for a smooth wall. In the laminarization regime the effect of roughness on heat transfer is negligible. Although the throat data for the $30^o$ nozzle may not agree with this hypothesis, the authors feel that this discrepancy is due to an instrumentation problem.

One of the consequences of accelerating flow is the phenomenon of laminarization. Therefore by decreasing the acceleration, as happens when going from the $60^o$ to $30^o$ nozzle, the Reynolds number range over which laminarization takes place should decrease. This is borne out in Fig. 6, where the 30° nozzle has fewer points in the laminarization region than the 60° nozzle. As a result the transition from laminarized flow in the 30° nozzle takes place at a lower Reynolds number than in the 60° nozzle.

For corresponding Reynolds numbers and roughness values the heat transfer parameter is greater for the 60° nozzle than for the 30°. In the 30° nozzle the axial length is greater and consequently the boundary layer is thicker at the corresponding diameters. As previously mentioned the thermal boundary layer acts as a conduction layer and the thicker it is the less heat transfer there is across it. Exceptions occur when the 60° nozzle heat transfer is in the laminarized regime while corresponding points for the 30° nozzle are in the transition or turbulent regimes.

A plot of heat transfer coefficient as a function of local nozzle diameter is presented in Fig. 7. Data from both nozzles are shown using an adiabatic inlet at a stagnation pressure, $p_o = 75$ psia. For the purpose of clarity, only the smooth and 325 rms wall surfaces are shown. The heat transfer distribution for the rough surface is typical, the 60° nozzle having a higher peak value of $h_i$ than the 30°. Although one would normally expect the same relationship of heat transfer between the two nozzles for the smooth wall as occurred for the rough wall, the results are to the contrary. The heat transfer coefficient for the 30° nozzle is considerably greater than that for the 60° nozzle at the stations in the vicinity of the throat. This is one of those exceptional cases that occur when the 60° nozzle heat transfer is in the laminarized regime while corresponding points for the 30° nozzle are in the transition regime.

## CONCLUSIONS

An experimental investigation has been performed in order to study the effects of roughness on heat transfer in conical nozzles. For a roughness range from smooth to 325 rms the following conclusions can be drawn:

1. There is a roughness height below which a nozzle can be considered to be hydraulically smooth.
2. Roughness causes detransition from turbulent flow to take place at a lower Reynolds number than that for a smooth wall.
3. In the laminarization regime, the effect of roughness on heat transfer is essentially negligible.
4. In the turbulent regime the heat transfer does not seem to be noticeably affected until the roughness exceeds a given height which corresponds to the approximated sublayer height.

## REFERENCES


1. Boldman, D. R., Schmidt, J. F., and Ehlers, R. C., "Effect of Uncooled Inlet Length and Nozzle Convergence Angle on the Turbulent Boundary Layer and Heat Transfer in Conical Nozzles Operating with Air," Journal of Heat Transfer, Vol. 89, No. 4, Nov. 1967, pp. 341-350.
2. Boldman, D. R., Neumann, H. E., and Schmidt, J. F., "Heat Transfer in 30° and 60° Half-Angle of Convergence Nozzles with Various Diameter Uncooled Pipe Inlets," TN D-4177, 1967, NASA, Cleveland, Ohio.
3. Graham, R. W. and Boldman, D. R., "The Use of Energy Thickness in Prediction of Throat Heat Transfer in Rocket Nozzles," TN D-5356, 1969, NASA, Cleveland, Ohio.
4. Back, L. H., Massier, P. F., and Cuffel, R. F., "Flow Phenomena and Convective Heat Transfer in a Conical Supersonic Nozzle," Journal of Spacecraft and Rockets, Vol. 4, No. 8, Aug. 1967, pp. 1040-1047.
5. Boldman, D. R., Schmidt, J. F., and Gallagher, A. K., "Laminarization of a Turbulent Boundary Layer as Observed from Heat-Transfer and Boundary-Layer Measurements in Conical Nozzles," TN D-4788, 1968, NASA, Cleveland, Ohio.
6. Reshotko, M., Boldman, D. R., and Ehlers, R. C., "Heat Transfer in a 60° Half-Angle of Convergence Nozzle with Various Degrees of Roughness," TN D-5887, 1970, NASA, Cleveland, Ohio
7. Nikuradse, J., "Laws of Flow in Rough Pipes," TM 1292, 1950, NACA, Washington, D.C.
8. Schlichting, H., Boundary Layer Theory, 4th ed., McGraw-Hill, New York, 1960.

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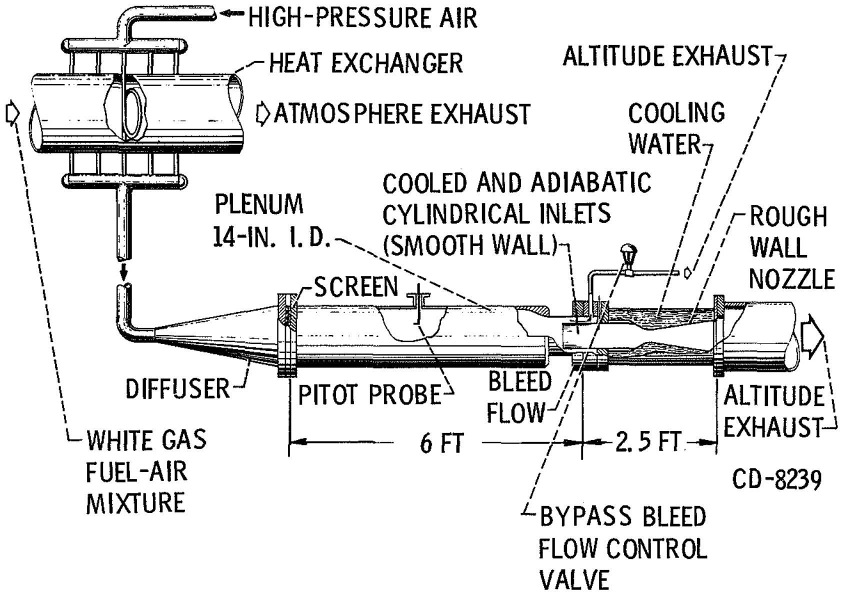
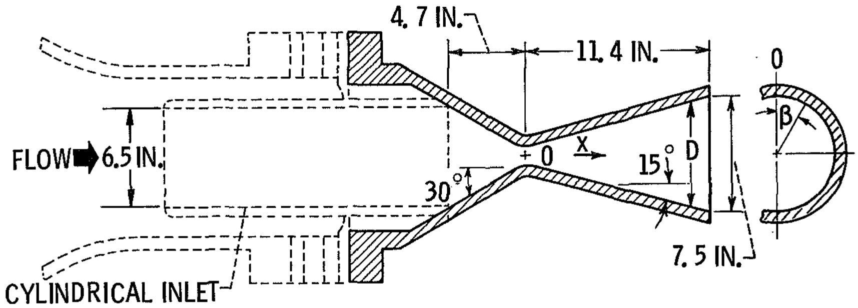



Figure 1. - Schematic diagram of nozzle heat-transfer facility.




| STATION | ANGULAR POSITION, β, DEG | | AXIAL DISTANCE, x, IN. | DIAMETER, D, IN. | AREA RATIO, $A/A_t$ |
|---|---|---|---|---|---|
| | PRES-SURE TAP | HEAT FLUX METER | | | |
| 2 | 231 | 51 | -4.515 | 6.250 | 17.548 |
| 3 | 129 | 309 | -3.515 | 5.092 | 11.648 |
| 4 | 180 | 0 | -2.512 | 3.934 | 6.952 |
| 5 | 231 | 51 | -2.158 | 3.528 | 5.591 |
| 6 | 283 | 103 | -1.812 | 3.128 | 4.395 |
| 7 | 334 | 154 | -1.460 | 2.722 | 3.328 |
| 8 | 26 | 206 | -1.110 | 2.316 | 2.410 |
| 9 | 77 | 257 | -.613 | 1.760 | 1.392 |
| 10 | 129 | 309 | -.175 | 1.510 | 1.024 |
| 11 | 180 | 0 | 0 | 1.492 | 1.000 |
| 12 | 231 | 51 | .130 | 1.502 | 1.013 |
| 13 | 283 | 103 | .255 | 1.540 | 1.065 |
| 14 | 334 | 154 | .392 | 1.604 | 1.156 |
| 15 | 26 | 206 | .634 | 1.732 | 1.348 |
| 16 | 77 | 257 | 1.221 | 2.042 | 1.873 |
| 17 | 180 | 0 | 2.736 | 2.858 | 3.669 |
| 18 | ↓ | ↓ | 5.468 | 4.322 | 8.391 |
| 19 | | | 8.201 | 5.792 | 15.070 |
| 20 | | | 11.032 | 7.320 | 24.071 |

Figure 2. - Instrumentation for 30° - 15° nozzle.

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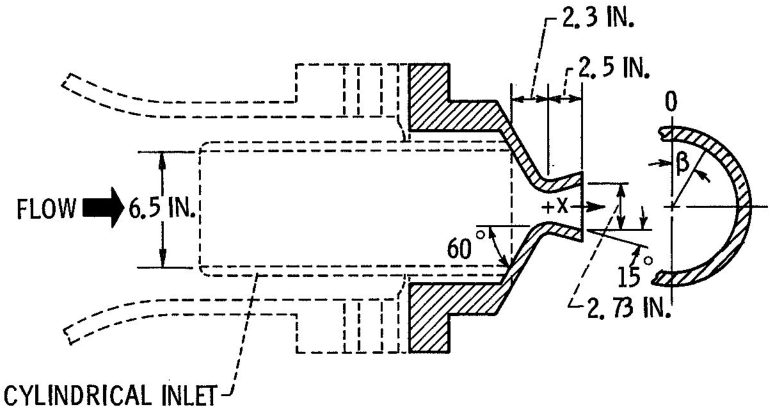


| STATION | ANGULAR POSITION, β, DEG | | AXIAL DISTANCE, x, IN. | DIAMETER, D, IN. | AREA RATIO, $A/A_t$ |
|---|---|---|---|---|---|
| | PRESSURE TAP | HEAT FLUX METER | | | |
| 2 | 180 | 0 | -2.085 | 5.740 | 14.663 |
| 3 | 77 | 257 | -1.752 | 4.584 | 9.352 |
| 4 | 129 | 309 | -1.635 | 4.178 | 7.769 |
| 5 | 180 | 0 | -1.515 | 3.762 | 6.299 |
| 6 | 231 | 51 | -1.206 | 2.732 | 3.322 |
| 7 | 283 | 103 | -1.020 | 2.309 | 2.373 |
| 8 | 334 | 154 | -.581 | 1.736 | 1.341 |
| 9 | 26 | 206 | -.146 | 1.516 | 1.023 |
| 10 | 77 | 257 | 0 | 1.499 | 1.000 |
| 11 | 129 | 309 | .150 | 1.521 | 1.030 |
| 12 | 180 | 0 | .277 | 1.565 | 1.090 |
| 13 | 231 | 51 | .400 | 1.623 | 1.172 |
| 14 | 283 | 103 | .622 | 1.741 | 1.349 |
| 15 | 334 | 154 | 1.209 | 2.053 | 1.876 |
| 16 | 26 | 206 | 2.140 | 2.545 | 2.883 |

Figure 3. - Instrumentation for 60° - 15° nozzle.

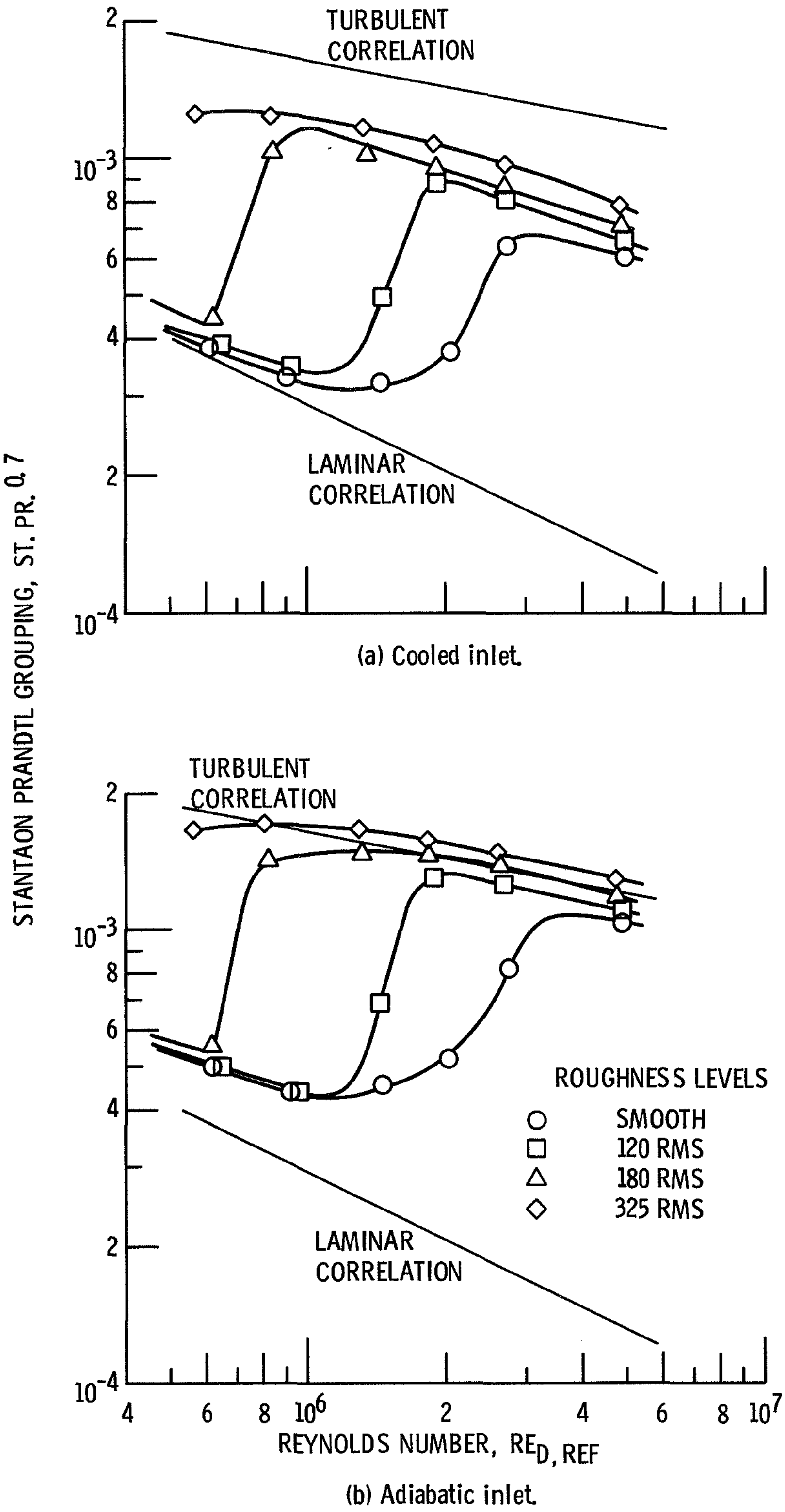


Figure 4. - Experimental heat transfer in the 60° half angle of convergence nozzle for various degrees of roughness at the throat station.

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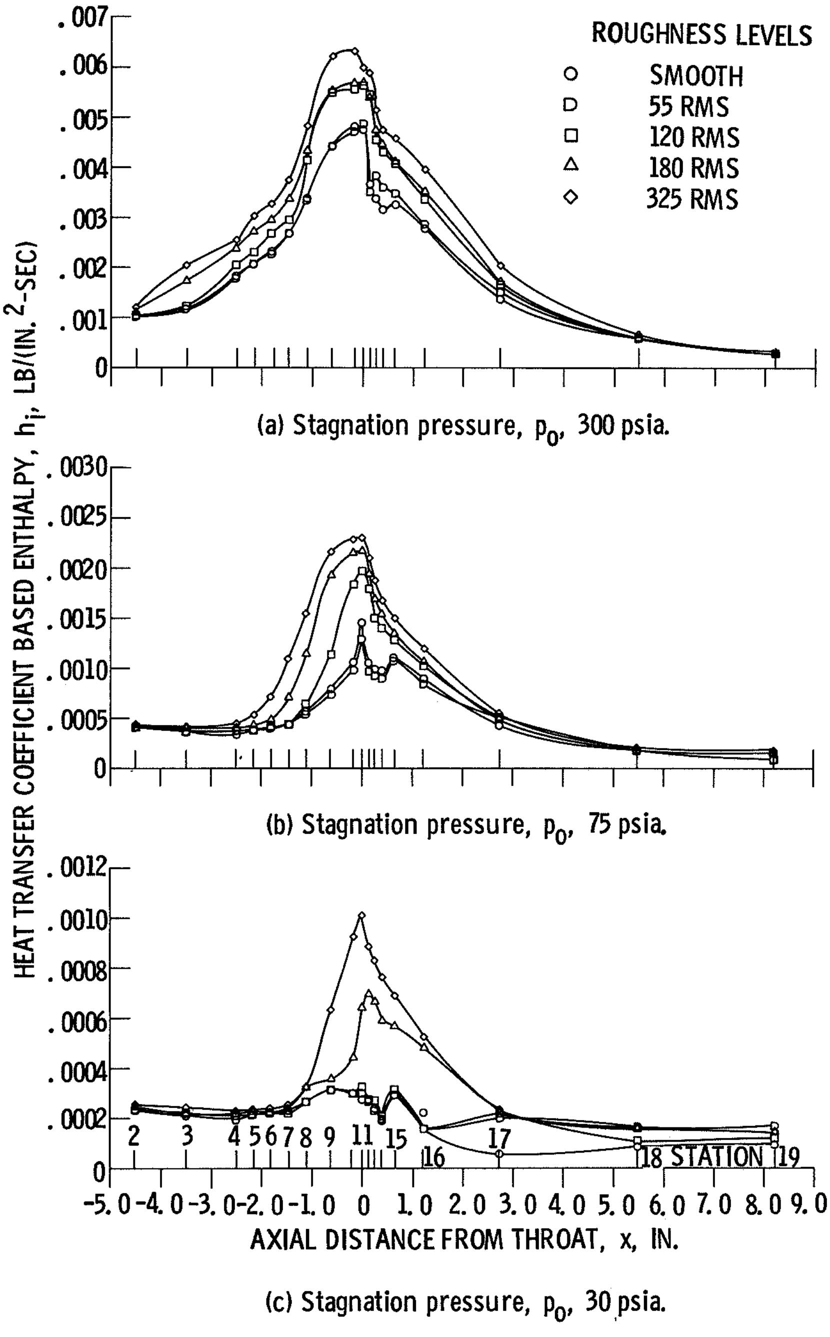
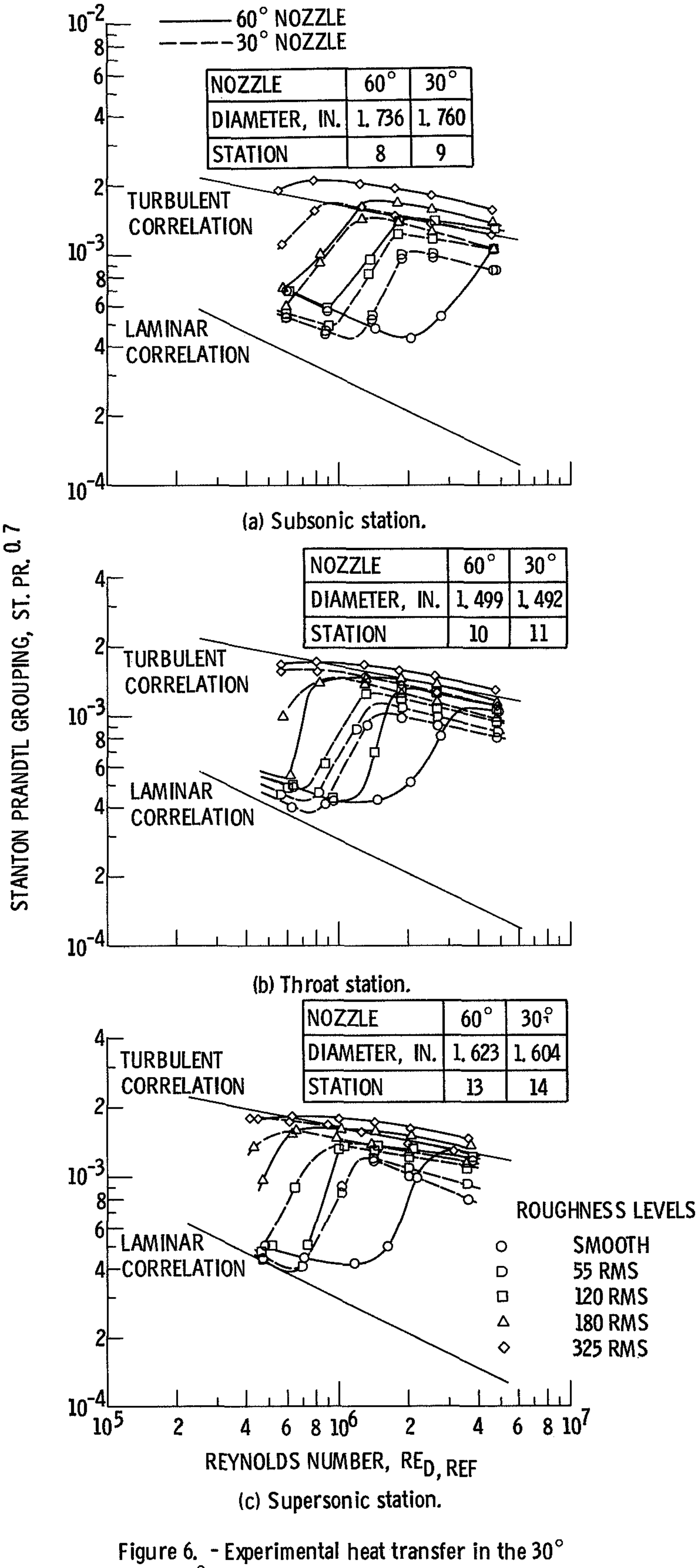



Figure 5. - Experimental heat transfer distribution in the 30° half angle of convergence nozzle for various degrees of roughness with an adiabatic inlet.




(c) Supersonic station.

Figure 6. - Experimental heat transfer in the 30° and 60° half angle of convergence nozzles for various degrees of roughness with an adiabatic inlet.

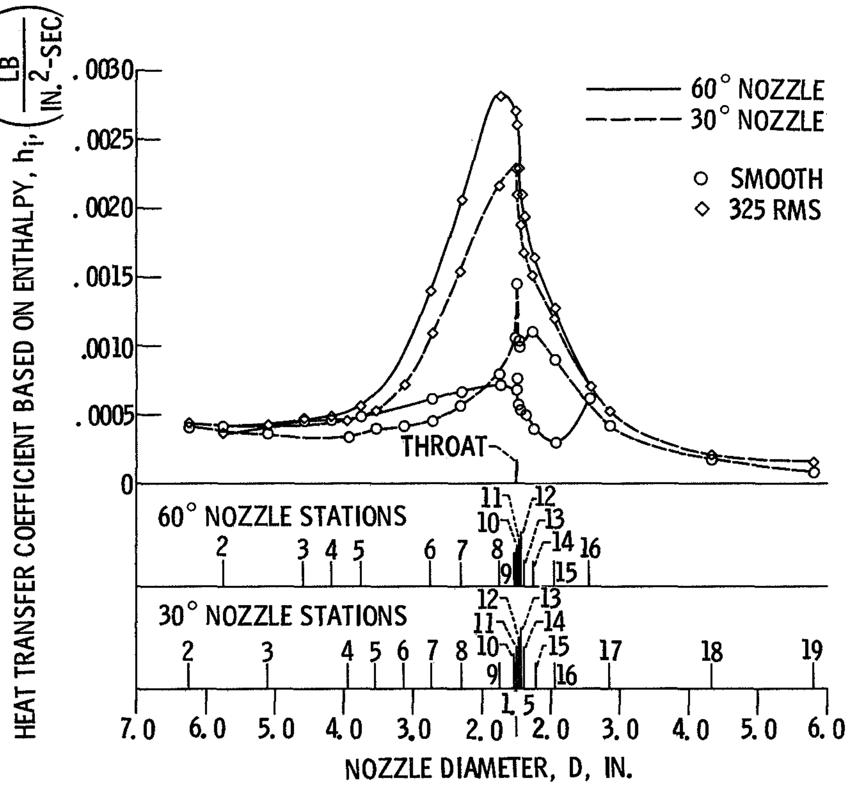



Figure 7. - Effects of roughness on heat transfer coefficient in a 30° and 60° half angle of convergence nozzle with an adiabatic inlet. Stagnation pressure, $p_0$, 75 psia.